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# CASE FILE

# SHOCK AND VIBRATION TESTS OF A SNAP-8 MERCURY PUMP

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### ABSTRACT

A SNAP-8 mercury pump was vibration and shock tested to the maximum levels anticipated during its usage. The tests included random vibration, sinusoidal sweep vibration, and multiple 15 G shocks. The mechanical and structural deficiencies exposed by the tests can be remedied with minor design changes.

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### SUMMARY

A SNAP-8 mercury pump was vibration and shock tested to the maximum levels anticipated during its usage. A detailed inspection of the pump after the tests revealed a loss of bearing preload on the angular-contact ball bearings. In spite of the preload loss, the bearing balls and race surfaces were in excellent condition. Both inner rings of the bearings were in good condition and came off the shaft assembly freely. Both of the outer rings of the bearings were tight in the housing bores and had to be pressed out. The housing bores were found to be galled.

The mechanical and structural deficiencies exposed by the tests were minor and some simple design changes are required in order to meet the vibration and shock requirements.

### INTRODUCTION

As part of the overall program for development of the SNAP-8 mercury Rankine space power system, an evaluation has been made of the capabilities of the major components of the power conversion system to meet or exceed the vibration and shock requirements. The test loads applied were the same as those typically imposed during launch and flight maneuvers. One of the components tested was an induction-motor-driven centrifugal pump designed for pumping mercury. The test operations were conducted in the environmental test laboratory at the Lewis Research Center. The disassembly and detailed inspection of the pump assembly were performed by the power conversion system contractor, Aerojet General Corporation, Azusa, California. This report covers a description of the testing and a post-test evaluation of the mercury pump.

### PUMP DESCRIPTION

The SNAP-8 mercury pump is shown mounted on the test shaker in figure 1 with a cutaway view shown in figure 2. The pump consists basically of a one-piece solid shaft supported by two preloaded angular-contact ball bearings mounted in a central housing. The bearings are lubricated with polyphenyl ether, and the mercury-oil sealing during operation is effected by visco and molecular pumps and by lift-off face seals when inoperative. A 208-volt 400-hertz three-phase induction motor is over-hung from the bearing housing at one end, and a combination jet-and-centrifugal pump is overhung at the other end. The jet pump is incorporated upstream of the centrifugal pump to suppress cavitation during startup and during zero-gravity operation. The pump has a mean diameter of 20 centimeters (8 in.), and the length is 76 centimeters (30 in.). The weight is approximately 68 kilograms (150 lb).

A more complete description of the pump may be found in reference 1.

### TEST FACILITY AND EQUIPMENT

A SNAP-8 mercury pump was vibration and shock tested to the maximum levels anticipated during its usage. The test operations were conducted in the environmental test laboratory at the Lewis Research Center. A sketch of the pump, showing accelerometer location and load-input axes, is presented in figure 3. For the shock tests the accelerometer outputs were fed into an oscilloscope and recorded photographically. For the vibration testing the accelerometer outputs were conditioned and recorded on G-level versus frequency plots.

The exciter for both the shock and vibration tests was an MB Electronics Corporation's "Electrodynamic System Model C210" (fig. 4). The exciter head may be used in the vertical position as shown (X axis in this test) or rotated counterclockwise  $90^{\rm O}$  and used in the horizontal position (Y and Z axis in this test).

The accelerometers used during this test were standard production units and included single-, dual-, and tri-axial units.

### TEST PROCEDURE

All the tests were conducted with the pump in a static, or noncperating, condition. All of the required connecting lines were capped as close to the pump as possible (fig. 1).

The mercury pump was vibration and shock tested to the maximum levels anticipated during its usage. Since the mounting configuration or orientation on the launch vehicle had not been established, the tests were conducted at maximum levels on each of three mutually perpendicular axes. The tests and test levels were as follows:

- (1) A sinusoidal vibration sweep test with a 1G input from 5 to 2000 hertz at a sweep rate of 1 octave per minute.
- (2) A random vibration test. The input levels were +3 decibels per octave (dB/oct) from 20 to 100 hertz, 0.4  $G^2$ /hertz from 100 to 600 hertz. and -6 dB/oct from 600 to 2000 hertz.
- (3) A 15G shock test in both the plus and minus directions, repeated three times in each axis (a total of 18 shocks). The wave shape was a half-sine pulse with 11 millisecond time duration.

Table I relates the test run, test axis input, input "G" levels, frequency, sweep speed, running time, and type of test.

### TEST RESULTS

The test data showed that during the X-axis sinusoidal vibration sweep, Accelerometer #14 (X axis on the shaft assembly) recorded a 29G peak response at 230 hertz (fig. 5). Accelerometer #15 (Z axis on the shaft assembly) recorded a 15 G peak response at 230 hertz (fig. 6). The data from Accelerometers #14 and #15 taken during this test are considered as typical, but they show the maximum recorded responses during this vibration sweep. Accelerometers #16 and #17 failed to function during this test.

The "Y" axis sinusoidal vibration test shows that both Acceler-ometers #14 and #15 recorded a 10G peak response at 150 hertz. Accelerometers #8 and #11 failed to function during this test.

The "Z" axis sinusoidal vibration test shows that Accelerometer #14 recorded a 7G response while Accelerometer #15 recorded a 10 G response, both at 140 hertz. Accelerometers #6 and #9 failed to function during this test.

During the first attempt to run the "X" axis sinusoidal vibration sweep test, the pump-mounting-cradle support pins loosened, and had to be retightened. The pump-mounting cradle and support pins are test-support items and are not the same as the mounting bracket that would be used during a full-system test. The entire "X" axis sinusoidal vibration sweep test was repeated.

### EXTERNAL EXAMINATION OF THE PUMP

A visual examination of the pump revealed a crack at the root weld on a 3.2-millimeter (1/8 in.) pressure tap in the pump suction line (fig. 7).

Upon removing the lubricant filter bracket (fig. 1) from the housing group, it was determined that the mounting bolt torques had decreased

from 1865 newton-centimeters (165 in.-1b) when installed to between 283 and 395 newton-centimeters (25 and 35 in.-1b) after the test. The other bolts in the housing group pattern which secures the motor stator were normal with 150 to 155 newton-centimeters (1470 to 1525 in.-1b) disassembly torque.

The pump-mounting-cradle support pins (fig. 1) which secured the main housing in the support cradle for this test were heavily galled on the pump side and less damaged on the motor side. The cradle and pins are test support fixtures and are not a part of the pump.

### INTERNAL EXAMINATION OF THE PUMP

A helium-leak check of the motor stator disclosed that the hermetically sealed terminals and thermocouple plug were leak tight to specification.

The mercury end of the pump was disassembled without any problems and all elements were in excellent conditions with no signs of damage. The lift-off seals, including carbon faces, were in original condition.

When the motor stator was removed, fine metallic particles were observed on the motor-cavity lubricant slinger and shroud. These came from an antirotation device which had been added to prevent rotation of the rotor during the test, and is not a part of the mercury pump. The tang on the aluminum antirotation device was battered from repeated heavy contacts with the locking grooves on the motor and bearing locknut. No damage was observed on the locking grooves on the locknut. The motor locknut had not loosened on its threads, but because of local yielding in the locknut-contact area, the disassembly torque was 68 newton-centimeters (6 in.-lb) compared to an assembly value of 2825 newton-centimeters (250 in.-lb). The motor had circumferential play on the shaft because of the axial looseness which was due to the local yielding in the locknut contact area. No significant damage was observed on the locknut, shaft threads, shaft key slot, key, or motor key slot.

The inner rings of both bearings came off the shaft freely and without damage, but both of the outer rings were tight in the housing bores and had to be pressed out. Under normal condition these outer rings are free to slide in the housing within the preloading limits. Numerous V-shaped shallow gouge marks were made in the housing surfaces during the bearing outer-ring removal (figs. 8 and 9). Out-of-roundness for the bearing seats in the housing was measured as 0.00020-centimeter (0.00008-in.) total indicator reading (T.I.R.) at the motor end, and 0.00025-centimeter (0.00010-in.) T.I.R. at the pump end. The high points did not match the damage pattern in a regular fashion. This was checked to eliminate the possibility of grinding chatter as a major contributor to the problem. The outside diameters of both bearings and both housing bores were inspected to locate any possible ridges

or sharp edges at the ends of tapers or corner radii, but none was found.

The housing had galled in several places and had deposited metal in a cold-welded fashion onto the bearing outer rings, keying them to the housing. These cold-welded deposits gouged the housing as the bearings were pressed out. The bearing stops showed no impact effects. The preload springs were undamaged and were within calibration limits.

### Bearings

Pump-end bearing. - The outer ring of the pump-end bearing (fig. 10) had nine large galled or material-pickup areas measuring approximately one millimeter by 3 millimeters (0.04 by 0.12 in.) each, and several small areas distributed around the circumference. All damage marks started at approximately 4 millimeters (0.16 in.) from the nonthrust face as chatter marks that extended across the ring towards the thrust face. This was caused by axial displacement of the outer race. Between the galls was a narrow circumferential band of disturbed finish which delineates a plane of contact around the periphery of the bearing. The outer raceway was free of all signs of vibration damage. A few very light raceway scratches appeared to be the result of efforts to remove the outer rings from the housing.

The raceway of the inner ring was undamaged. Extremely light ball tracks were attributed to noise testing of the bearing as part of the manufacturer's acceptance tests. Light axial mounting marks were seen in the bore. The balls were entirely free of any damage or marking. The separator (fig. 11) showed a light ball contact pattern in the pockets and a light contact pattern on the outside diameter.

Motor-end bearing. - The outer ring of the motor-end bearing (fig. 12) had 17 galled areas of various sizes distributed around the outer diameter. The galled areas started at approximately 4.8 millimeters (0.19 in.) from the nonthrust face. In line with the galls was a light narrow circumferential band of disturbed finish which appeared to have been generated by repeated contact with the housing bore. The raceway had very light ball-contact patterns at 0.26 and 0 radian (150 and 00) contact angles with normal ball spacing. The nonthrust portion of the raceway had minor scratches, probably from the disassembly efforts.

The inner ring showed two or three incomplete sets of ball-contact patterns at 0.30 and 0 radian ( $17^{\circ}$  and  $0^{\circ}$ ) contact angles. These were at normal ball spacing and showed contact with three to five balls at each set. The appearance was that the inner ring and ball orientation shifted many times during the testing.

The balls were in excellent condition. The separator (fig. 13) showed a light ball-contact pattern in the pockets and a light contact pattern on the outside diameter.

### DISCUSSION

The greater amount of bearing and housing damage occurred at the motor end as expected since the overhung motor mass is largely supported by the motor-end bearing. The pump-end bearing is additionally protected by the carbon seals which provide damping of any motion at the pump end. The low  $[0 \text{ radian } (0^0)]$  contact-angle pattern found in the motor-end bearing indicated a complete loss of preload during a portion of the testing. The loss of bearing preload permitted large displacements of the shaft and explained the damage to the antirotation device. The reorientation of the balls and the bearings rings would not have occurred if the preload had been continuously applied.

The pressure tap that cracked at the weld, and similar pressure taps on other lines were cantilever mounted with relatively large-mass closure cups on the ends. The large mass caused excessive vibration amplitudes and resulted in this weld crack. The pressure taps were provided for development testing and would probably be eliminated from a flight system or, if retained, would be better supported.

The loosening of the motor on the shaft and the loss of torque on the adjacent locknut are related problems. First, the motor had an excessively long drive key which, even though properly fitted, would bear on only a few high points. It would take a large force to seat a key that was closely fitted, and this might result in an eccentric position of the rotor on the shaft, which would be detrimental to rotor balance; this eccentric position is permitted by a loose fit between the rotor internal diameter and the shaft outer diameter, diametral clearance being from 0.0125 to 0.050 millimeter (0.0005 to 0.0020 in.). To reduce rotor runout, it is necessary to fit the key on the loose side. Use of a standard ball-bearing locknut and washer tended to further aggravate the situation. Axial clamping would hold the rotor against rotation if sufficient torque were applied. However, any circumferential motion tends to wear down the high points on the soft lockwasher, reducing the clamping force. This is a common problem and may occur in other locations where such lockwashers are used.

### CONCLUDING REMARKS

- 1. The loss of bearing preload can be considered as a possible potential failure since subsequent successful operation of the unit would be questionable. This could be corrected by redesign of the bearing housing to employ hardened steel liners or hard chroming the housing surfaces to improve sliding of the bearing outer rings.
- 2. The motor loosening and loss of torque on the locknut are a minor problem since function was not affected.
- 3. Small unsupported pressure taps cannot survive the vibration testing.

4. For the SNAP-8 mercury pump, some minor design changes are required in order to meet the required vibration and shock specifications.

### REFERENCES

1. Pope, J. R.; Mah, C. S.; and Foss, C. L.: Design and Development of a SNAP-8 Mercury Pump Motor. NASA CR-72664, 1970.

TABLE I

Type test	Run number	Axis test	"G" level input	Frequency input, hertz	Sweep speed	Running time
Sine	1	X	1	5-2000	1 oct/min	8.4 min
Sine	9	Y	1	5-2000	1 oct/min	8.4 min
Sine	10	Z	1	5-2000	1 oct/min	8.4 min
Random	2	Х	3 db/oct 0.4 g <sup>2</sup> /hertz -6 db/oct	20-100 100-600 600-2000	20 g's overall	3 min
Random	18	Y	iı	11	11	11
Random	11	Z	11	11	11	11
Shock Shock	3,4,5 6,7,8	+X -X	15 15			11 ms 11 ms
Shock	19,20,21	+Y	15			8 ms
Shock	22,23,24	-Y	15			8 ms
Shock	12,13,14	+Z	15			8 ms
Shock	15,16,17	<b>-</b> Z	15			8 ms

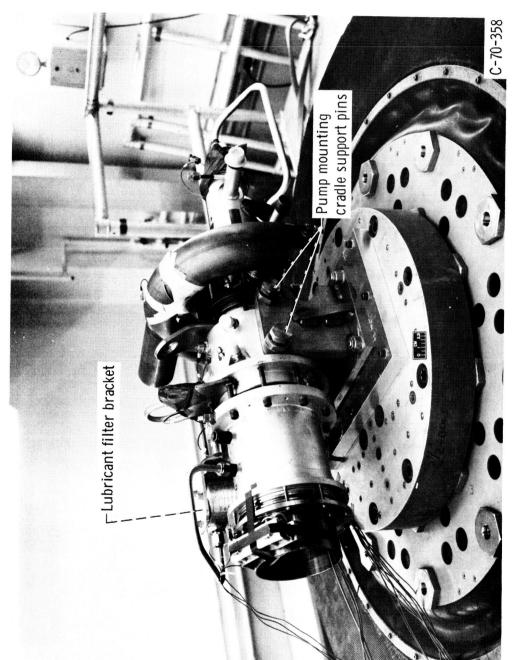


Figure 1. - SNAP-8 mercury pump.

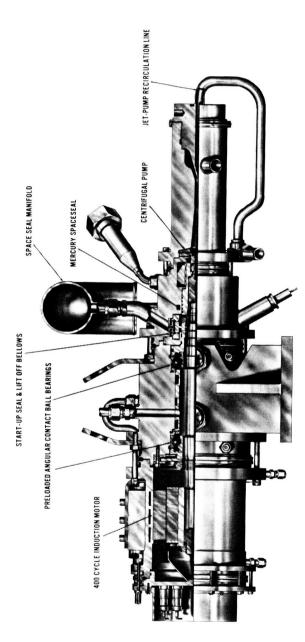


Figure 2. - SNAP-8 mercury pump cross-section.

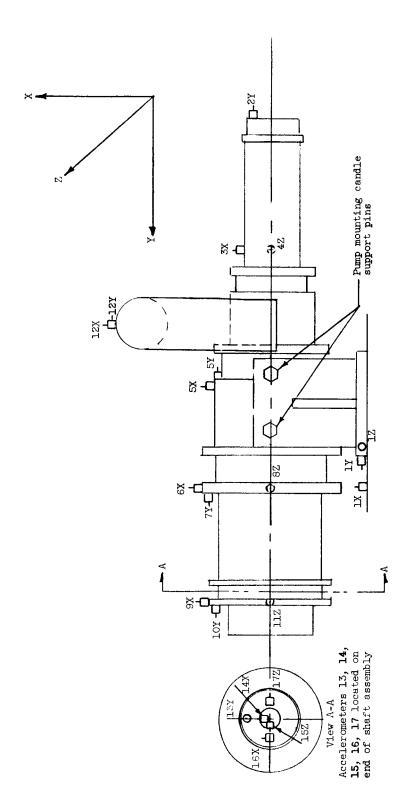


Figure 3. - Accelerometer locations and vibration axes.

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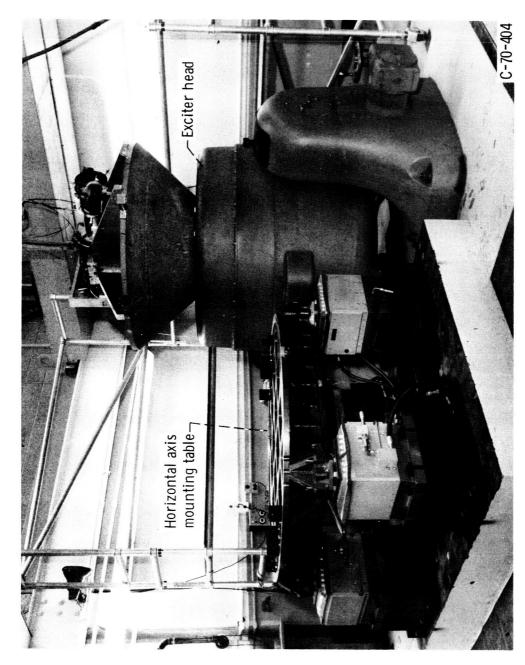


Figure 4. - Vibration and shock machine MB electronics, electrodynamic system model C 210.

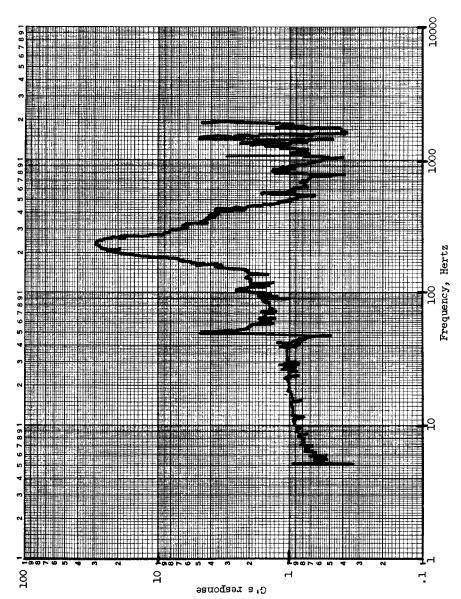
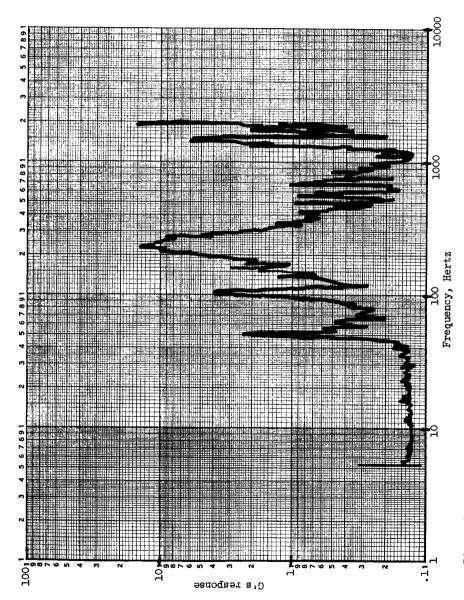


Figure 5. - Typical response of X axis accelerometer #14 to 1G X axis sine input for octave/min sweep.



axis accelerometer #15 to 1G X axis sine input with Ŋ  $_{
m Jo}$ Figure 6. - Typical response octave/min sweep.

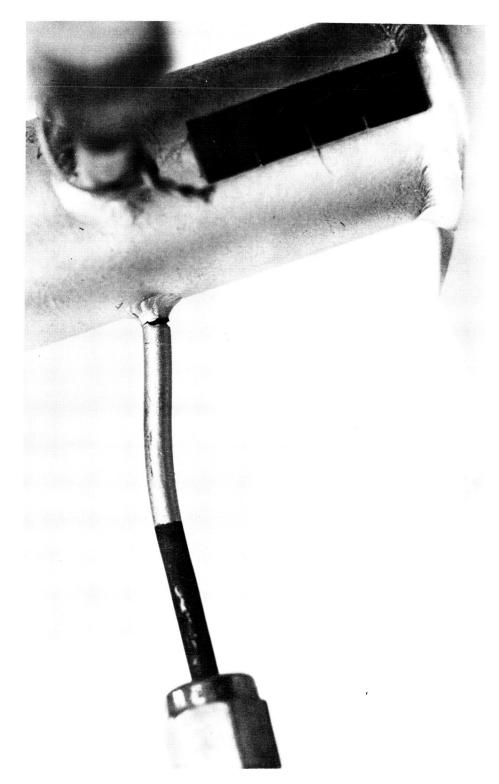


Figure 7. - SNAP-8 mercury pump after vibration testing. Pump housing showing crack at weld of pump suction line, pressure tap.



Figure 8. - SNAP-8 mercury pump after vibration testing. Transmission housing showing damage to pump end bearing seat.

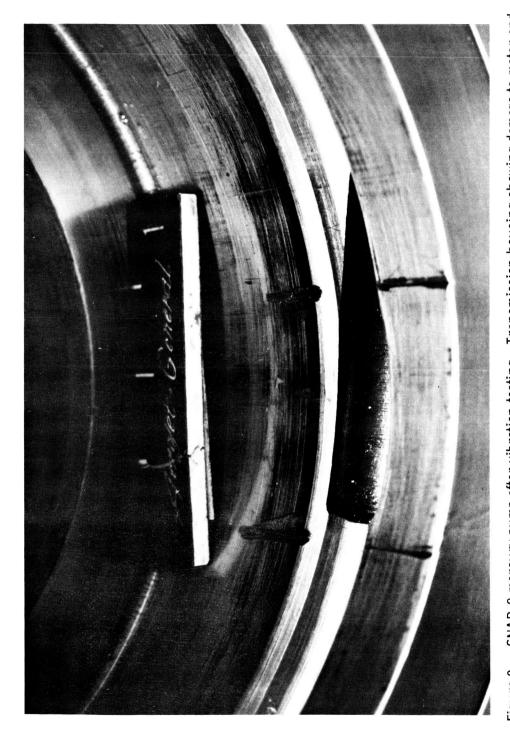


Figure 9. - SNAP-8 mercury pump after vibration testing. Transmission housing showing damage to motor end bearing seat.

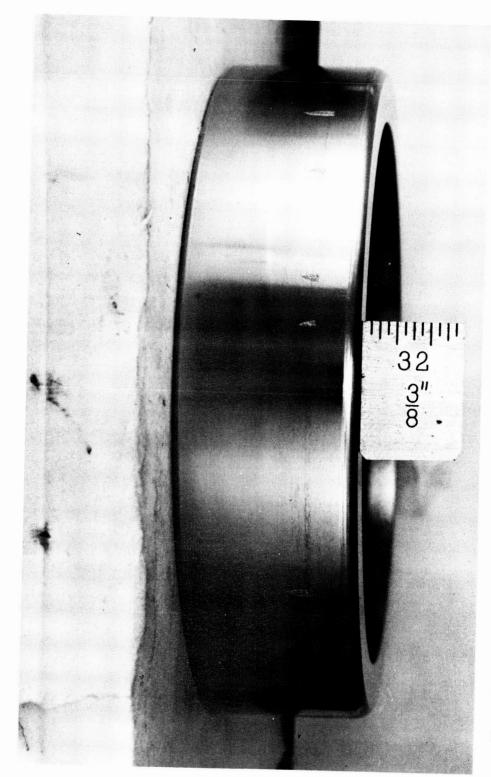


Figure 10. - SNAP-8 mercury pump after vibration testing. Pump end bearing outer ring showing galls and scratches.

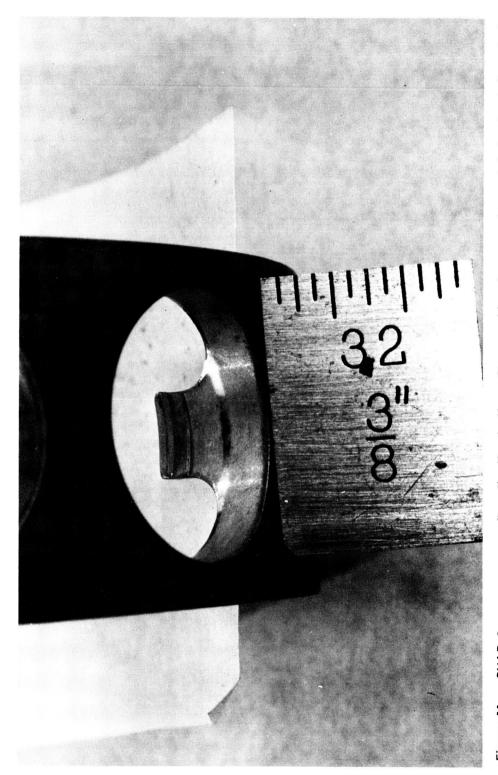


Figure 11. - SNAP-8 mercury pump after vibration testing. Pump end bearing separator pocket showing ball contact pattern.

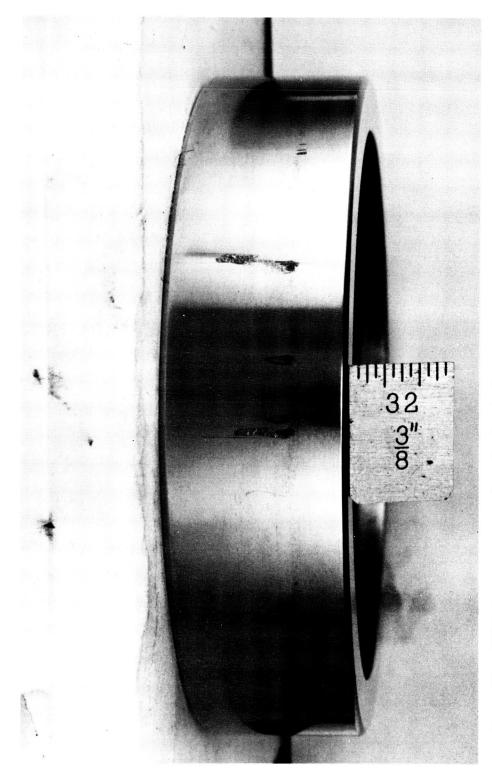


Figure 12. - SNAP-8 mercury pump after vibration testing. Motor end bearing outer ring showing galls and scratches.

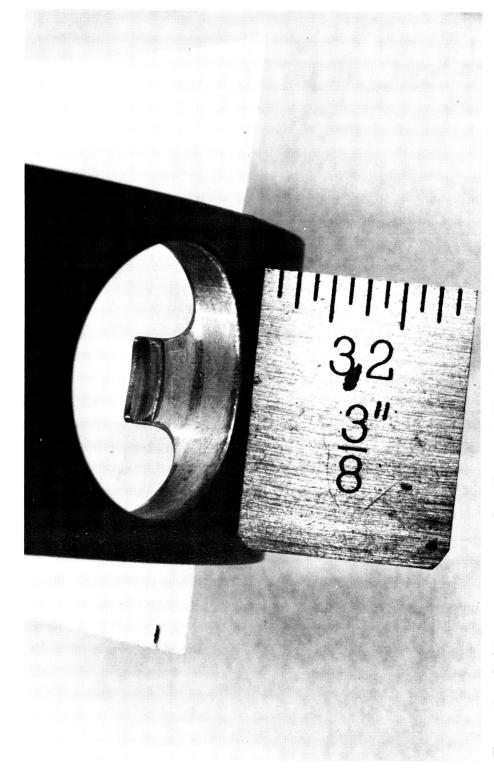


Figure 13. - SNAP-8 mercury pump after vibration testing. Motor end bearing separator pocket showing ball contact pattern.